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Bristow et al.

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[54] **POSITIVE DISPLACEMENT PUMP SYSTEMS**

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[52] **U.S. Cl.** **417/302**; 417/308

[58] **Field of Search** 417/300, 302, 303, 304, 417/308

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[57] **ABSTRACT**

To minimize the power absorbed by a positive displacement pump system used where the requirement for pressure fluid varies inversely with the pump speed, notably in automobiles, the system provides two separate delivery passages **61, 62** for the pumped fluid and a discharge passage **66** into which the fluid from the delivery passage is passed under the control of a valve means **70, 71,** the valve means commencing to operate on one **(62)** of the two delivered flows at lower speeds to by-pass a proportion of said one flow to an overspill **79,** while the fluid not bypassed is added to the flow from the other delivery passage being passed to the discharge passage. The said proportion is increased in a manner to maintain the pressure drop across a discharge orifice in the discharge passage 66 at a constant value, and as the pump speed increases further the valve means commences to by-pass also an increasing proportion of the flow from said other delivery passage 61 to the overspill to maintain the said pressure drop constant. The increase in area of communication between the second delivery passage and overspill is greater than the increase in area of communication between the first delivery passage and overspill. In this way, the final regulation is carried out on a smaller quantity of fluid and less fluid is pumped to the highest pressure in the system.

21 Claims, 5 Drawing Figures

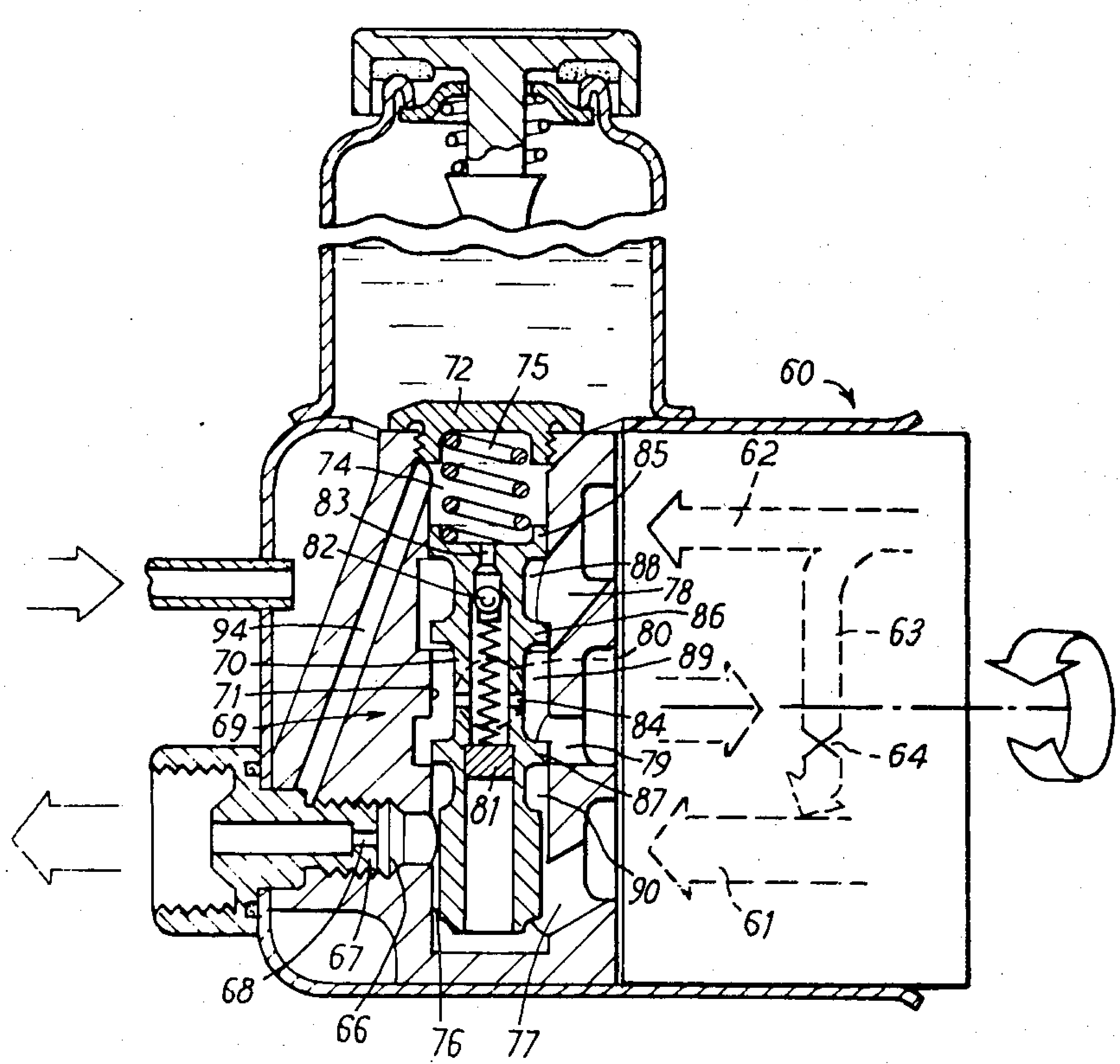


FIG. 1

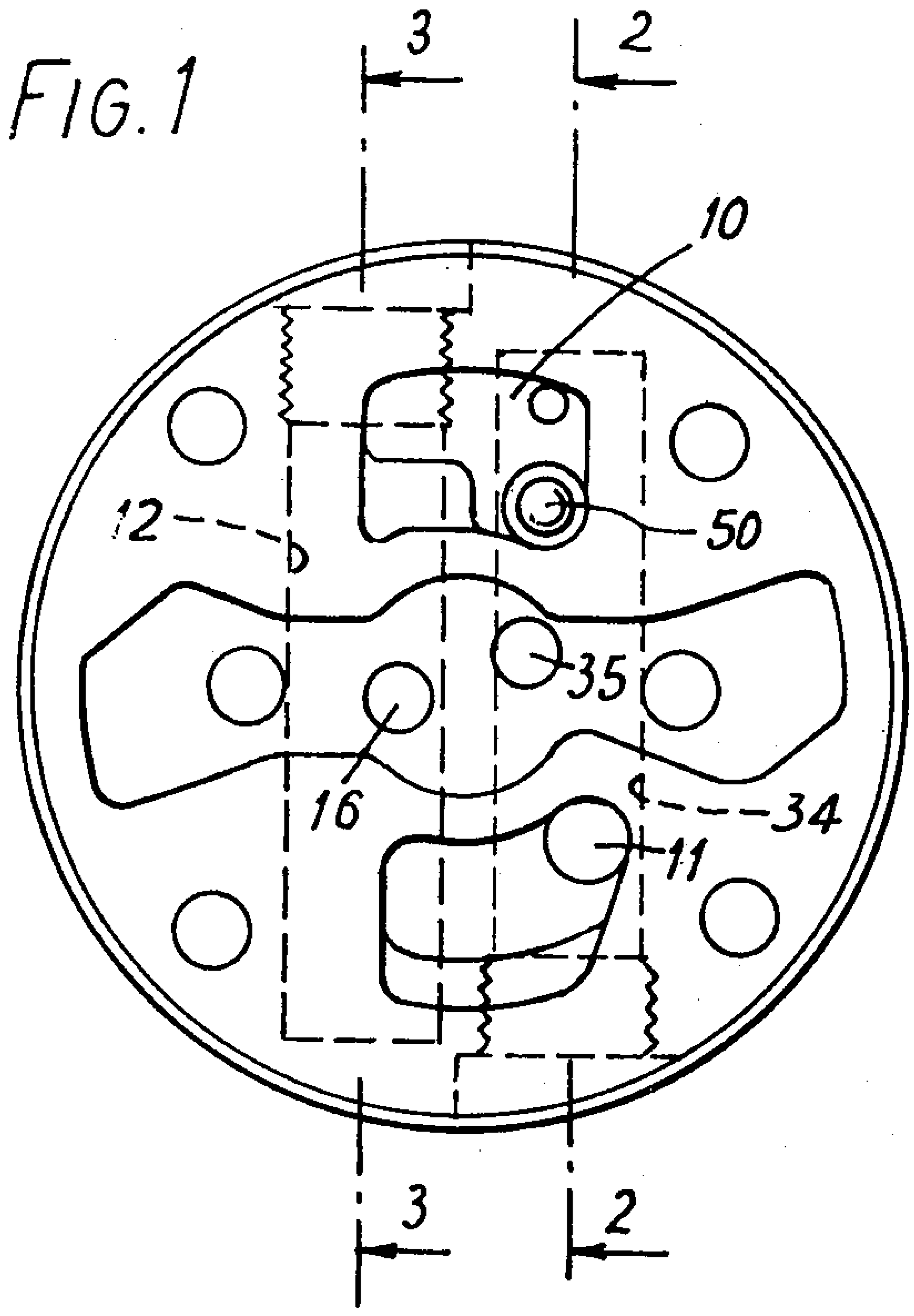


FIG. 2

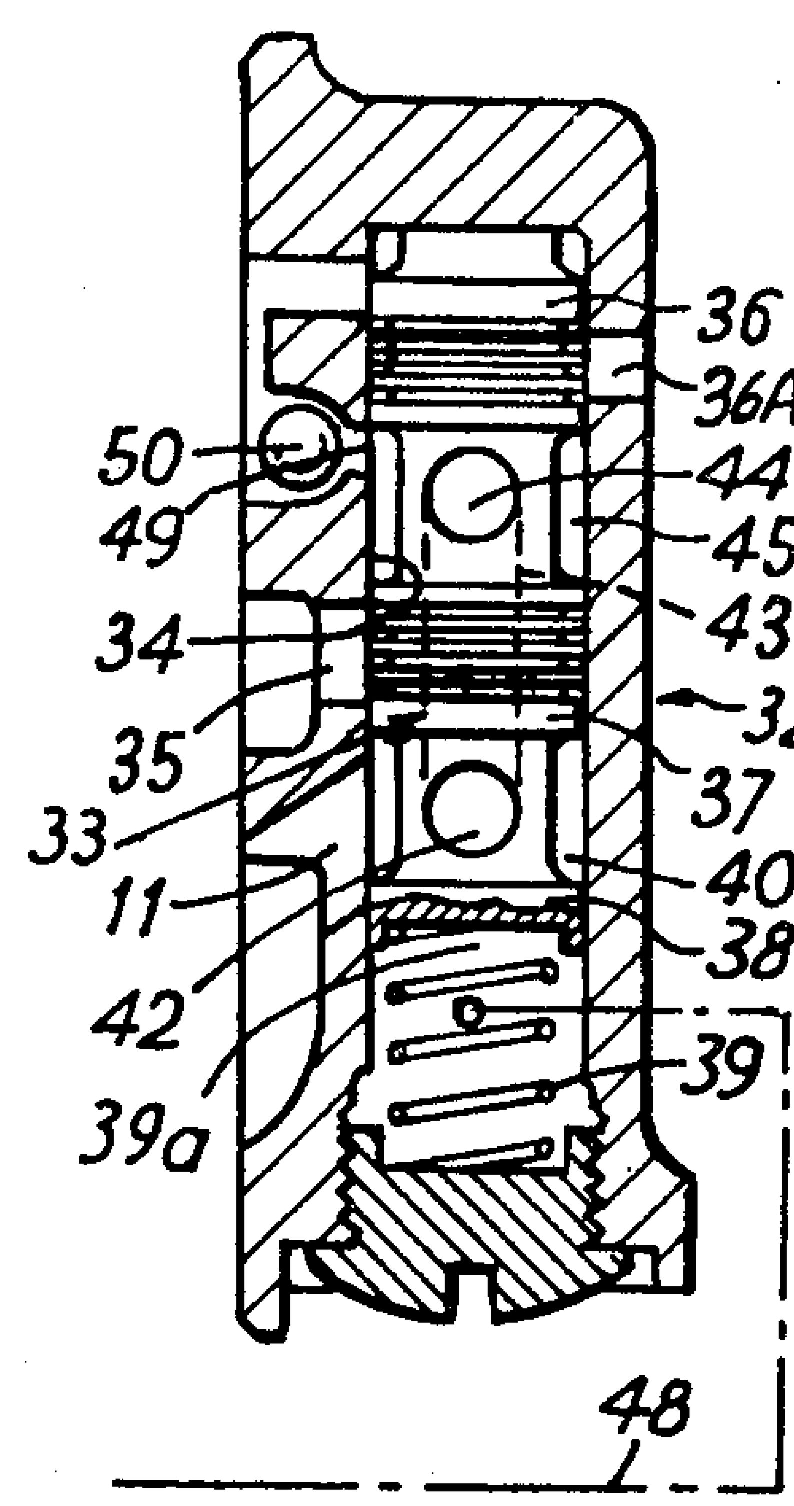


FIG. 3

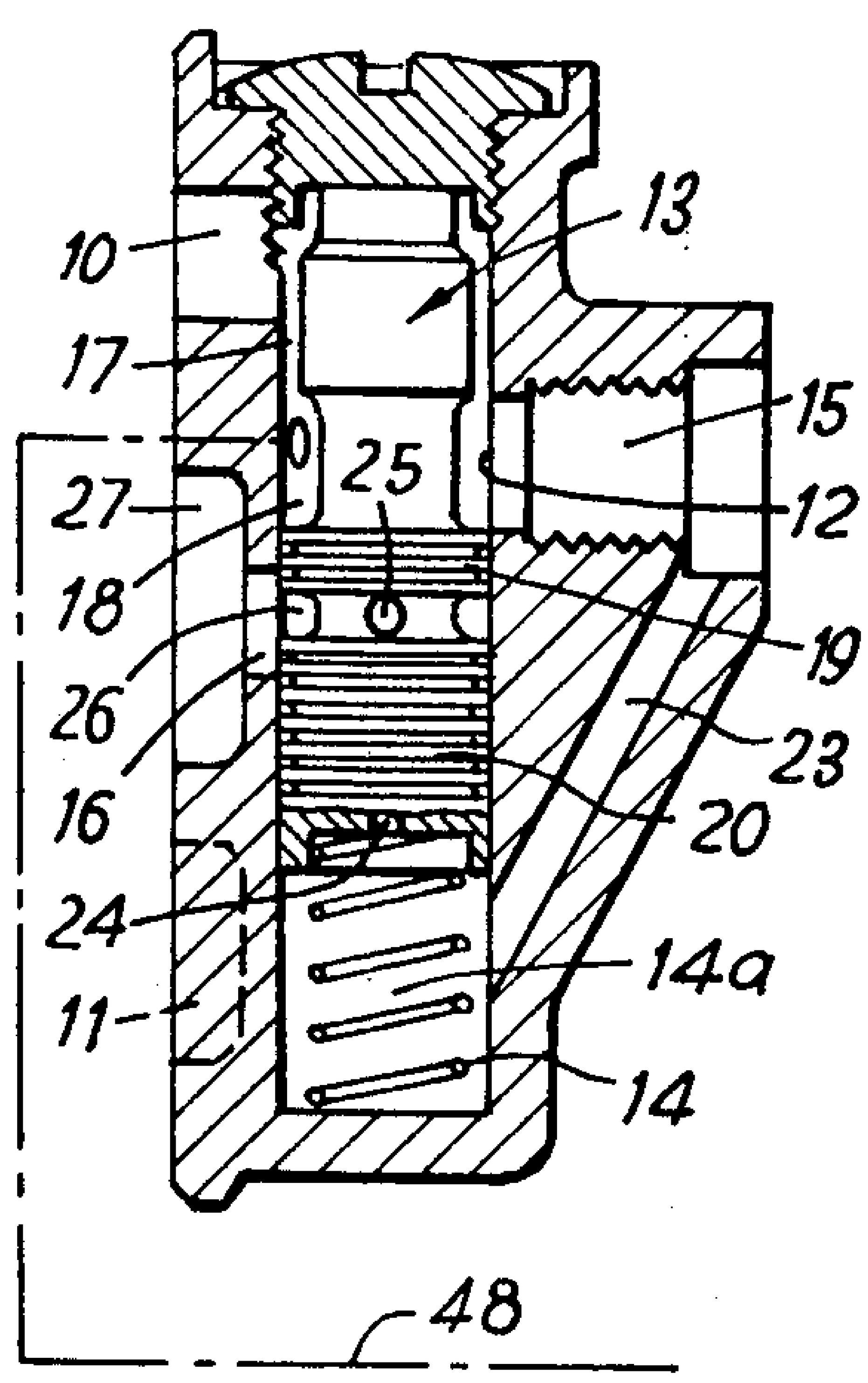


FIG. 4

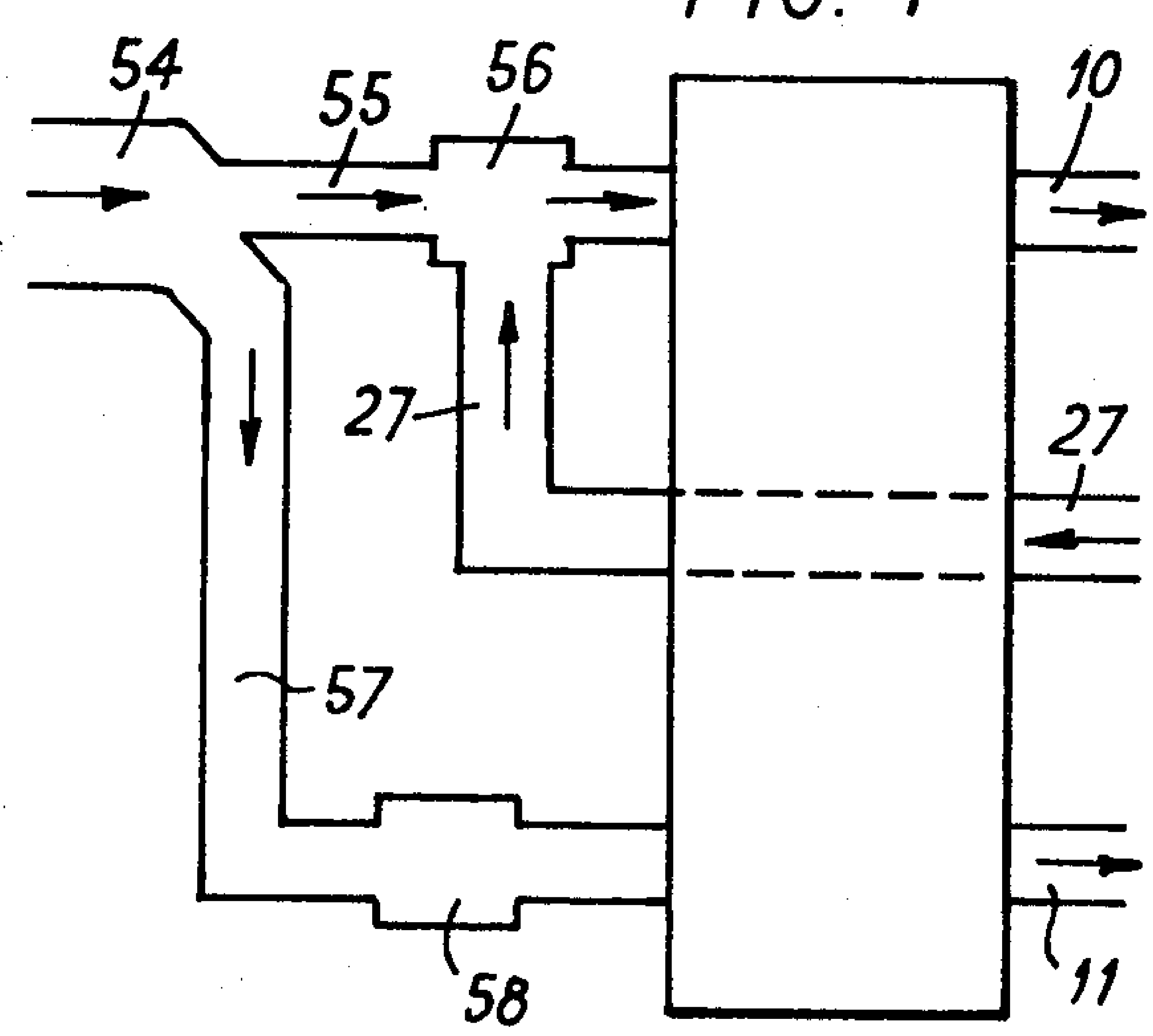
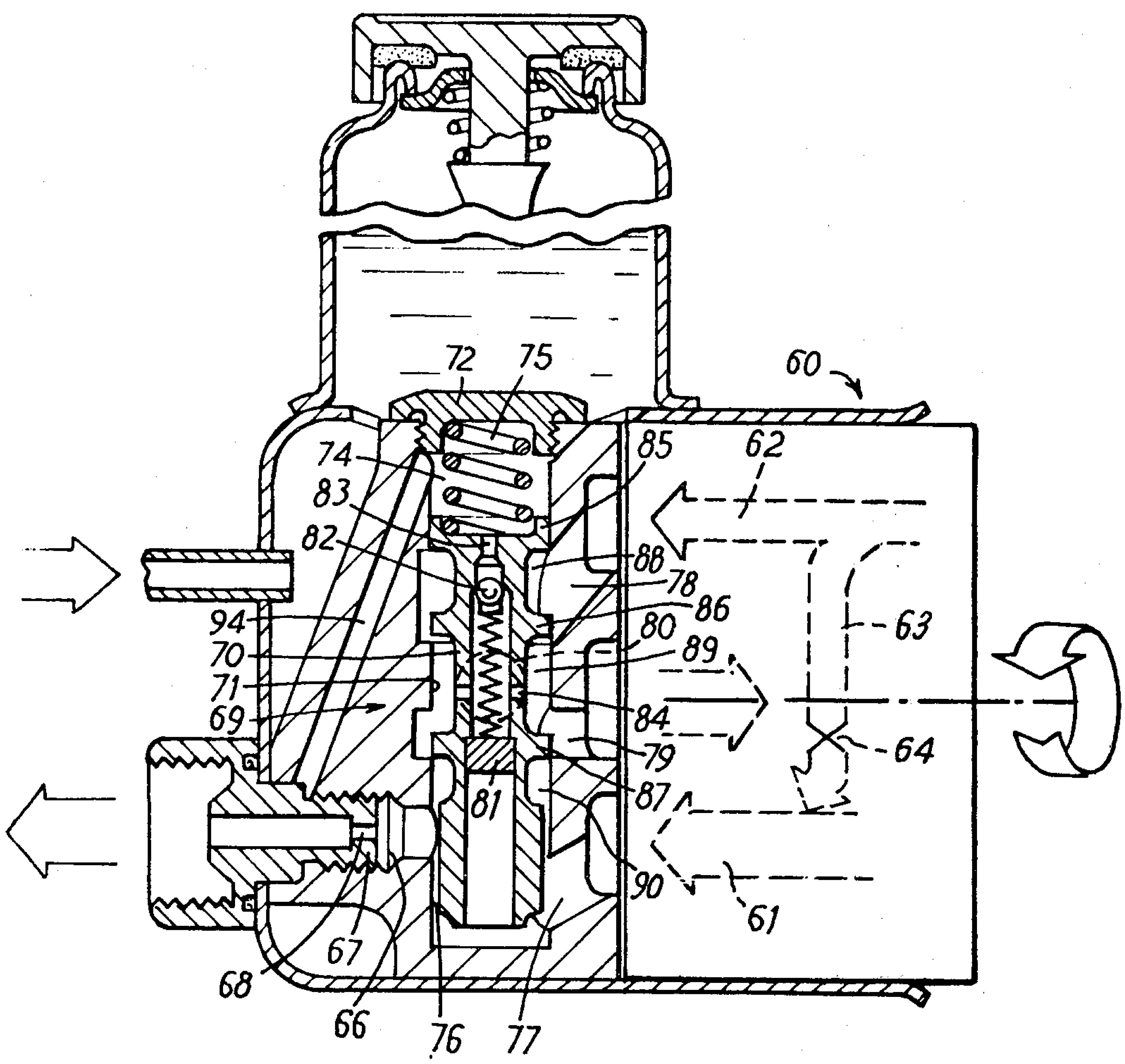


FIG.5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POSITIVE DISPLACEMENT PUMP SYSTEMS

This invention relates to positive displacement pump systems equipped with valve means for controlling the delivery of the pump. Such valve means may for example control the quantity of liquid pumped to an external circuit substantially constant whatever the pump speed, or may maintain the pressure in an external circuit up to a predetermined maximum pressure. A pump system incorporating the former type of valve means has an important but not exclusive application in power-assisted steering arrangements for motor vehicles.

According to this invention there is provided a positive displacement pump system having first and second delivery passages for first and second flows of pumped fluid respectively, a main discharge passage for pumped fluid, overspill ducting, and valve means comprising a control valve controlling the apportionment of the first flow between the main discharge passage and the overspill ducting as a function of the delivery pressure of the first flow in a sense to increase the proportion of the first flow by-passed to the overspill ducting as said pressure increases and to decrease the proportion of the first flow by-passed to the overspill ducting as said pressure decreases, a transfer passage through which fluid can flow from the second delivery passage to join the first flow, and a transfer valve controlling, as a function of the delivery pressure of the first flow, the apportionment of the second flow between the overspill ducting and said transfer passage the proportion of the second flow by-passed to the overspill ducting increasing with increase of the delivery pressure of the first flow and decreasing with decrease of the delivery pressure of the first flow.

The control valve and the transfer valve may each comprise a valve member rotatable relative to a plate so as to control ports to which the overspill ducting and the appropriate passage or passages through which the flow of fluid controlled by the valve passes. In the embodiment more particularly described hereinafter the control valve and the transfer valve are each however in the form of a slide member mounted in a bore to which the passage or passages and the overspill ducting open. In preferred arrangements the control valve member and the transfer valve member are constituted by different portions of a common movable member.

Said function of the delivery pressure of the first flow may be the delivery pressure itself or a pressure directly proportional to the delivery pressure, but particularly useful embodiments, for use in supplying servo fluid at a constant rate to opencentre servo valves of power-assisted steering arrangements, can be obtained by employing, as said function, the pressure drop across an orifice in the main discharge passage. The pressures upstream and downstream of the orifice can conveniently be applied directly on a valve member of the slide type to urge the valve member in opposite directions.

Some embodiments of the invention will now be described by way of example. The description makes reference to the accompanying diagrammatic drawings in which:

FIG. **1** shows the ported face of an end member of a pump system embodying the invention,

FIGS. **2** and **3** are respectively sectional views on the plane **2**—**2** and **3**—**3** of FIG. **1**,

FIG. **4** is a diagram of a preferred inlet passage system for use in pump systems according to the invention, and

FIG. **5** is a diagrammatic sectional elevation of the end member of a preferred system according to the invention.

The invention is applicable to any type of positive displacement pump system which has at least two outlet ports which are respectively supplied with fluid from separate pump chambers, but is particularly advantageous in pumps in which it is possible to arrange that a greater amount of fluid can be delivered to one of the ports than to the other. One such pump is the vane pump comprising an axially central generally annular member of which the radially inner surface is a cam profile and is engaged by vanes carried in respective generally radially extending slots in the periphery of a rotor which is secured to one end of a driving shaft, and end members secured to axially opposite sides of the central member. One of the end members provides a bearing for supporting the driving shaft and affords also an inlet passage leading by way of two branch passages to two inlet ports opening to the face of the end member adjacent the central member. The cam profile is such as to cause each vane to move radially inward and outward twice in each revolution of the rotor so that each vane performs two pumping cycles in each revolution. By shaping the cam profile so that one of the radial inward and outward movements is greater than the other, one of the pumping cycles can be caused to produce a greater delivery than the other. Two delivery passages receiving the pumped fluid from the two inlet ports respectively lead to respective delivery ports formed in the other end member and are in communication with each other under the control of a valve means interconnecting the outlet ports, a main outlet passage for the pumped fluid, and overspill porting. A pump of this type is described and shown in our British Patent Specification No. 818,644.

When the pump is used to provide a flow of servo fluid for power assisted steering in a motor vehicle, the requirement for a high pressure output occurs chiefly when the vehicle is being parked or is turning sharp corners, for both of which maneuvers the vehicle speed and engine speed are normally relatively low. The output of the pump is greatest when the vehicle is travelling at high speed, at which time large movements of the steering wheel are not made and would indeed be dangerous. Because of these conflicting requirements for a high output at low speed and low output at high speed, it is usual to provide a valve means which, when the pump delivery is in excess of the requirements of the servo system, by-passes a proportion of the pumped fluid back either to the servo fluid reservoir or to the pump inlet passage.

Referring now to FIGS. **1** to **3** of the accompanying drawings the end member of the casing of a first pump embodying the invention is shown, which casing provides the two delivery ports for the pumped fluid. These ports are shown at **10** and **11** and will be referred to as the first and second delivery ports respectively.

The delivery port **10** opens to one end of a control valve bore **12** (see FIG. 3) in which a control valve member **13** is slidably mounted. The member **13** is urged towards the said one end of the bore by a compression spring **14** in a spring chamber **14***a*. Also opening to the bore **12** are the main outlet passage **15** for the pumped fluid and an overspill port **16**. In flowing from the first delivery port **10** to the main outlet passage **15**, the fluid passes through an annular restricted passage **17** between the valve member and the wall of the bore, into an annular chamber **18** about the valve member and thence into the outlet passage. The valve member has two lands **19, 20** which close off communication between chamber **18** and the spring chamber **14***a* respectively and the overspill port **16**. A constant-flow control orifice (not shown) is disposed in the outlet passage **15** and a pressure-sensing passage **23** extends from the downstream side of this orifice to the spring chamber **14***a*. An axial passage **24** extends through the valve member from the spring chamber to a cross-bore **25** opening to an annular space **26** which is formed between lands **19** and **20** of the valve member and which is in permanently open communication with the overspill port **16**. The passage **24** contains a pilot relief valve (not shown) which operates to relieve excessive pressure in the pump, resulting for example from excessive resistance to steering. This excessive resistance tends to produce a high pressure at the downstream side of the control orifice and hence in the spring chamber **14***a*. If the pressure in the spring chamber exceeds its maximum permissible safe value, the fluid pressure opens the relief valve, permitting fluid to escape from the spring chamber through cross-bore **25** to the overspill port. The resulting fall in the pressure in the spring chamber causes the valve member **13** to move so as to increase the area of communication between chamber **18** and overspill port **16** so that an increased amount of fluid from delivery port **10** is diverted from the outlet passage **15** to the overspill port, so as to prevent the maximum safe pressure in the spring chamber from being exceeded. The overspill port leads to a recess **27** which may communicate with the main inlet passage in the other end member of the pump, e.g. by way of an axial passage extending along the pump shaft, or with the reservoir.

The ports **10** and **11** are in communication with each other only through a transfer valve **32**, see FIG. **2**.

The transfer valve comprises a valve member **33** slidably mounted in a valve bore **34** which is closed at both ends and which is parallel to the control valve bore **12**. The valve bore **34** has an overspill port **35** which leads to the recess **27**.

The transfer valve member has three lands **36, 37, 38** and is loaded by a spring **39** in a chamber **39***a* into abutment with the end of the valve bore adjacent the first delivery port **10**. Between lands **36** and **37**, the valve member has a cross-bore **44**, and between lands **37** and **38** has a cross-bore **42**, the two cross-bores being connected together by an axial bore **43**.

In the initial or rest position of the transfer valve, land **36** is disposed adjacent the first delivery port **10** but permits pumped fluid from the second delivery port **11** to flow by way of an annular space **40** between lands **37, 38**, cross-bore **42**, axial bore **43** and cross-bore **44** in the valve member, an annular space **45** between lands **36** and **37**, and a gap indicated at **49** between land **36** and the edge of port **10**, to the first delivery port **10**. The overspill port **35** is blanked off by land **37**.

At low pump speed this flow from the second delivery port **11** thus joins the fluid delivered directly to the port **10** by the vanes and the combined flow moves along the restricted annular passage **17** to chamber **18** and thence to the outlet passage **15**.

A passage indicated by a chain line **48** in FIGS. **2** and **3** extends from the chamber **18** to the spring chamber **39***a*.

The operation of the valve is as follows:

At low pump speeds the spring **39** holds the transfer valve member in the position shown in FIG. **2**, so that all the pressure fluid from the secondary delivery port **11** flows through the cross-bore **42**, axial passage **43**, cross-bore **44** and annular space **45** to the delivery port **10**, and thence through the restricted passage **17** and chamber **18** to the main outlet passage **15**.

As the pump speed increases, the flow past the restriction **17** increases and causes an increasing pressure difference between port **10** and chamber **18**. Passage **48** extending from chamber **18** to the spring chamber **39***a* of the transfer valve operates as a pressure sensing passage, so that the pressure in chamber **18** is applied in chamber **39***a* whilst the pressure at port **10** is applied to the adjacent end of valve member **33**. The increasing pressure difference causes the transfer valve member **33** to move to reduce the size of the gap **49** between the land **36** and the edge of the port **10** and subsequently to commence to open communication between the annular passage **45** and the overspill port **35**, so that the flow from port **11** is apportioned between port **10** and the overspill port.

As the pump speed continues to rise, the increased flow through the restricted annular passage **17** produces a further increasing pressure difference between port **10** and chambers **18** and **39***a*, and the resulting movement of the transfer valve member **33** against the spring **39** is arranged to cause the hydraulic forces on the valve member to be such that it moves rapidly to increase the area of communication between the annular space **45** and the overspill port **35** while reducing the area of the gap **49**. The effect of this is to reduce the speed range over which the pressure in the second delivery port **11** is greater than that in the first delivery port **10**.

The increasing flow through the discharge orifice in the outlet passage **15** produces a lowering of pressure at the downstream side of the orifice, and this lower pressure is transmitted via the passage **23** to the spring chamber **14***a*. When the resulting pressure difference acting on the control valve member **13** rises sufficiently to overcome the force of the spring **14** holding the flow control valve in its closed position, the member will move and land **19** will commence to uncover the overspill port **16** and a proportion of the flow will thus be returned to the pump inlet passage or the reservoir as previously described.

In order to avoid a continuously increasing pressure drop through the restricted passage **17**, a hole **36A** in the bore of the transfer valve becomes partly uncovered by the land **36** and serves to return a proportion of the flow to the reservoir or to the pump inlet passage. Thus the amount of fluid flowing through the restricted passage **17** is also controlled.

Continuing increase of the pump speed increases the delivery by the pump to the first delivery port **10** to such an extent that the delivery to the second delivery port **11** becomes unnecessary, and it is arranged that the land **36** closes the gap **49** and that the effective area of communication between the annular space **45** and the overspill port **35** is so large that the pressure of the fluid in the delivery port **11** is very low, and consequently the power absorbed by the pump now increases at a lesser rate than before with increasing pump speed.

It is preferred to incorporate a non-return valve **50** in a branch of port **10** in the transfer valve member leading to gap **49** to avoid any reversal of flow from port **10** through the gap **49** to the overspill port **35**, such as may occur when the pump is running at low speed and is required to produce fluid at high pressure.

The restriction of passage **17** may be replaced by a restriction in the part of the first delivery port **10** which communicates with the control valve bore.

In another alternative design the pressure drop across the discharge orifice in the main outlet passage can be used instead of the pressure drop through the restricted passage **17**. In this case the preloading of the spring **39** is adjusted so as to control the flow from the pump until the quantity of fluid delivered by the pump directly into the first delivery port **10** is sufficient to allow the fluid delivered by the pump to the delivery port **11** to be passed in its entirety through the overspill port **35**.

As illustrated diagrammatically in FIG. **4**, the main intake passage **54** is preferably branched into a duct **55** leading to a first chamber **56** whence the fluid is pumped by the pump to the first delivery port **10**, and another duct **57** leading to a second inlet chamber **58** whence the fluid is delivered by the pump to the second delivery port **11**, and all the fluid from the overspill recess **27** is discharged into the said first chamber **56**. In this way the intake **55** to the section of the pump from which fluid is pumped to the first delivery port **10** is supercharged so as to improve the performance of the pump at high speed and prevent the onset of cavitation in this section of the pump.

Referring now to FIG. **5** of the drawings, a preferred pump according to the invention is illustrated in which the transfer valve and control valve of FIGS. **1** to **3** are combined, but the mode of operation is similar. The pump, shown diagrammatically at **60**, is required to deliver pressure fluid to first and second delivery passages **61**, **62** which are in communication with each other through a connecting passage **63** in the pump casing, and passage **63** contains a non-return valve, indicated at **64**, which permits flow from passage **62** to passage **61** but not in the reverse direction. The combined flow from passages **61** and **62**, less any which is surplus to the immediate requirements of the external circuit and which is directed to an overspill port **79** in the valve and thence to a fluid reservoir or the pump inlet for recirculation, is delivered to the external circuit through a main discharge passage **66** in which is mounted a threaded plug **67** providing a discharge control orifice **68**. The orifice is of accurately predetermined diameter according to the required fluid delivery, and the pressure drop across the orifice is applied to the combined valve **69** for the purpose of maintaining the flow through the orifice substantially constant.

The valve **69** comprises a valve member **70** slidably mounted in a valve bore **71**. The upper end of the valve bore has screwed into it a sealing plug **72** forming a chamber **74** at the upper end of the bore. Chamber **74** contains a spring **75** which urges the valve member **70** downward into abutment with the other end of the valve bore.

A continuation of the second delivery passage **62** beyond passage **63** communicates with the valve bore through a port **78**. The combined flow from passages **61** and **63** flows into the lower end of the valve bore through a port **77** and thence through an annular restriction **76** between the wall of the valve bore and a reduced-diameter end portion of the valve member into the discharge passage **66**. Between the port **78** and **77**, an overspill part **79** and an auxiliary overspill port **80** lead off the valve bore to the inlet passage system of the pump.

An axial bore extending along the valve member **70** contains a sealing plug **81** which serves as a base for a spring of a pilot relief valve **82** for relieving excess pressure in the spring chamber **74** via an axial hole **83**, radial holes **84** and the overspill ports **79**, **80**.

The external surface of the valve member has three axially-spaced annular lands **85**, **86**, **87** forming between them annular chambers **88**, **89**. A third annular chamber **90** extends about the valve member between the land **87** and the reduced diameter lower end portion of the valve member. Ports **79** and **80** together ensure that the annular chamber **89** is in communication with the overspill passage in all positions of the valve member.

Land **85** blanks off the spring chamber **74** from the port **78** in all positions of the valve member.

The valve member is shown displaced from its initial position at the lower end of the valve bore. In the initial position of the valve member, when the pump is operating at low speed, lands **86** and **87** blank off communication between port **78** and annular chamber **90** respectively and annular chamber **89**, and in consequence the full flow from the second delivery passage **62** flows through passage **63** to join the flow through the first delivery passage **61**. The combined flow passes through the annular restriction **76** and thence through the discharge orifice **68** to the external circuit.

The pressure at the downstream side of the discharge orifice **68** is applied to the upper end face of the valve member in the spring chamber **74** through a passage **94**. The pressure of the fluid acting against the lower end face of the valve member is augmented by the pressure at the upstream side of the orifice applied, in chamber **90**, to an annular area of the valve member equal to the area of the restriction **76** in a section plane at right angles to the axial dimension of the valve member. The force applied to the valve member against the force of spring **75** is increased by the presence of the restriction **76**, which causes a pressure slightly higher than the pressure in chamber **90** to be applied to the lower end face of the valve member. As the pump speed increases, the pressure drop across the orifice **68** also increases and when the pump delivery increases to a predetermined value the resultant force on the valve member overcomes the spring force and raises the valve member causing the land **86** to open communication between port **78** and chamber **89**, enabling some of the fluid from the second delivery passage **62** to flow through the overspill ports **79**, **80** so as to reduce the flow through the connecting passage **63**. As the pump speed continues to increase, the increasing pressure drops across restriction **76** and discharge orifice **68** cause the valve member to be lifted higher, and since the pressure drop across the restriction **76** increases according to substantially a square law the higher force applied in raising the valve member assists in overcoming Bernoulli forces, which resist the opening of cummunication between chambers **90** and **89**. The raising of the valve member opening communication between chambers **90** and **89** permits some of the flow issuing through port **77** to pass to the overspill. To pass a given amount of excess fluid from chamber **90** to overspill port **79** a certain axial movement of the valve member is necessary. This axial movement increases the area of opening between land **86** and the adjacent co-operating edge in the valve bore by a greater amount than that by which the area of communication between chamber **90** and port **79** is increased. As the amount of fluid to be passed through this area between land **86** and the co-operating edge in the valve bore does not rise by the same proportion, with increasing pump speed the axial displacement of the valve member reaches a value at which the fluid pressure in port **78** falls below that in port **77**, and at this stage the non-return valve **64** closes. As the pump speed increases further the flow control is exercised by land **87** on the flow from the first delivery passage **61**, while the proportionately greater increase in area of opening between land **86** and the adjacent co-operating edge in the valve bore causes a substantial reduction in the pressure in the second delivery passage **62**. At this stage, the cycle of the pump which is delivering fluid to the second delivery passage **62** is absorbing much less energy than if all the delivered fluid flowed through port **77** and the control were exercised by land **87**.

It will be appreciated that the control on flow from port **78** to the overspill port exercised by land **86** and the adjacent co-operating edge in the valve bore could be exercised alternatively by axial grooving in the valve member.

If the pressure at the downstream side of the discharge orifice **68** exceeds a predetermined maximum safe value, the pilot relief valve opens and allows fluid from the spring chamber to flow into the annular chamber **89** and thence to overspill. The resulting drop in the pressure in chamber **74** causes a corresponding upward movement of the valve member and increases the amount of fluid flowing to overspill from the delivery passages and thus operates to leave the pressure at the downstream side of the discharge orifice at a safe value.

The arrangement described in relation to FIG. **4** can advantageously be applied to the pump shown in FIG. **5**.

In some constructions intended for use other than in power-assisted steering it may be desirable to design the system so that fluid from the first delivery passage commences to pass to the overspill ports before fluid from the second delivery passage. Nevertheless, even in these arrangements, the fact that the rate of increase of area of communication between passage **62** and the overspill ports is greater than that of passage **61** leads to a greater fall in pressure in the passage **62** than in passage **61** over the normal range of movement of the valve member.

Where the pump is of a design in which the pressures in the two delivery ducts act in diametrically opposite directions on the pump rotor, the passing of the entire delivery from the second delivery passage to overspill at low pressure tends to unbalance the pump, and the unbalance may become acute at high pump speeds and may adversely affect the pump bearings if the delivery pressure is high. This problem can be alleviated to a substantial extent in the arrangement of FIG. **5** by replacing the nonreturn valve **64** by an orifice which restricts the flow between the first and second delivery passages to some extent. At low pump speeds, such a pump system operates in the same way as described above in relation to FIG. **5**, but when the pump speed increases to a value such that the fluid in the second delivery passage is being discharged to overspill at a lower pressure than obtains in the first delivery passage, the flow through the orifice is reversed so as to bleed a quantity of fluid from the first delivery passage to the overspill by way of the second delivery passage and to reduce the pressure in the first delivery passage at the higher pump speeds. The amount of fluid passing through the main discharge passage is still controlled by the valve in dependence on the pressure drop across the orifice **68**, and the valve is thus still automatically adjusted to deliver the same constant flow of fluid to the discharge passage. There is however a more progressive relief of the pressure in the first delivery passage with increase in pump speed.

In view of the saving of energy which results, at high pump speeds, from passing to overspill the whole of the delivery of the pump to the second delivery passage, it is advantageous in some cases to arrange for the pump to deliver a greater quantity of fluid to passage **62** than to passage **61**, so that at high speeds there is a correspondingly increased energy saving.

The pump shown in the drawings has numerous advantages, and permits other advantages to be obtained by appropriate design according to the purpose for which the pump is required, as follows:

(1) Since at high pump speeds the whole of the flow delivered to the second delivery passage is by-passed from the external circuit and is directed at low pressure into the overspill, substantial energy is saved at these higher pump speeds; furthermore, since this overspill liquid is not pumped to a high pressure its temperature remains at a lower value, which is advantageous in itself because it leads to a lower mean temperature of the body of working fluid in the pump system, but which leads to the further advantage that leakage from the pump is reduced, enabling a smaller pump to be used, and this in turn leads potentially to a saving in manufacturing costs and to a further saving of energy.

(2) Since the whole of the flow from one of the two delivery passages is, at high pump speeds, by-passed from the extenal circuit and passed on to an overspill passage, it can be advantageous to use a pump in which unequal quantities of pumped fluid are delivered to the two delivery passages, and in some important applications of the pump the second delivery passage may have the larger quantity of fluid pumped into it, with consequent increased energy saving at higher pump speeds.

(3) The amount of fluid fed to the final flow control section of the valve is much smaller, enabling improved regulation by the valve to be obtained.

(4) It is possible to obtain, plotting pump speed against delivery into the main discharge passage, a flat or falling characteristic, that is to say a characteristic in which the flow into the main discharge passage, having reached a maximum value at a given pump speed is maintained constant or decreases as the pump speed increases.

(5) Where, as in the case of a pump for power-assisted steering, the requirements is for a constant flow in the external circuit, additional pressure is created in the pump delivery for the purpose of moving the valve member to achieve the required control. This additional pressure inevitably introduces losses, but in the present constructions these losses are reduced because a lesser quantity of fluid is pumped to these pressures.

(6) By supplying fluid from the overspill to the pump inlet ports associated with the first delivery passage, improved filling of the pumping chambers in the relevant cycle is improved, and since that reduces the amount of noise emitted from the pump at high speed, the pump can run at higher speeds for a given permissible noise level.

(7) In an arrangement in which the pump is designed to deliver less fluid to the first delivery passage than to the second delivery passage, the former cycle of the pump can operate more satisfactorily at high speed since the amount of fluid to be drawn into the pumping

chambers is then less, and since the pump can thus run at a higher speed, a smaller pump can be used.

We claim:

1. A positive displacement pump system having first and second delivery passages for first and second flows of pumped fluid respectively, a main discharge passage for pumped fluid, overspill ducting, and valve means comprising a control valve controlling the apportionment of the first flow between the main discharge passage and the overspill ducting as a function of the delivery pressure of the first flow in a sense to increase the proportion of the first flow by-passed to the overspill ducting as said pressure increases and to decrease the proportion of the first flow by-passed to the overspill ducting as said pressure decreases, a transfer passage through which fluid can flow from the second delivery passage to join the first flow, said value means further comprising a transfer valve controlling, as a function of the delivery pressure of the first flow, the apportionment of the second flow between the overspill ducting and said transfer passage, the proportion of the second flow by-passed to the overspill ducting increasing with increase of the delivery pressure of the first flow and decreasing with decrease of the delivery pressure of the first flow.

2. A pump system as claimed in claim **1**, wherein the valve means comprises a piston valve member slidably mounted in a valve bore having lands and co-operating with said bore to carry out the functions of both the control valve and the transfer valve.

3. A pump system as claimed in claim **2**, wherein the piston valve member is loaded in one axial direction by a spring disposed in a spring chamber, and wherein an entry port for the first fluid flow and a main outlet port open to the valve bore and are in permanently open communication with each other, a pressure of the fluid flowing through the said entry port being applied to the valve member in opposition to the force of the spring.

4. A pump system as claimed in claim **3**, comprising a restrictor through which the fluid from the main outlet port is passed, and a duct extending between the downstream side of the restrictor and said spring chamber.

5. A pump system as claimed in claim **4**, in which said second delivery passage has a continuation extending from said transfer passage and opening to said valve bore, and said overspill porting opening to the overspill ducting from a location in the valve bore axially spaced from the continuation, said piston valve member being axially movable against the spring force from an initial position in which the piston valve member blanks off communication between the continuation and the overspill porting into an off-loading position in which the piston valve member opens communication between the continuation and the overspill porting.

6. A pump system as claimed in claim **5**, wherein the piston valve member has a portion which blanks off communication between the entry port for the first fluid flow and the overspill porting, and said pressure of the fluid which flows through the entry port urges the piston valve member in a direction progressively to move said portion to bring the entry port into communication with the overspill porting.

7. A pump system as claimed in claim **6**, wherein the overspill porting is sized and disposed in co-operation with the piston valve member and the continuation to cause each movement of the piston valve member in opposition to the spring force, from commencement of communication between the entry port and the overspill porting, to increase the area of communication between the continuation and the overspill porting by a greater amount than that by which the area of communication between the entry port and the overspill porting is increased.

8. A pump system as claimed in claim **3**, further comprising a spring-loaded relief valve for limiting the pressure in the spring chamber.

9. A pump system as claimed in claim **8**, wherein the relief valve is mounted in a passage extending axially along the piston valve member, which passage is in permanently open communication with the overspill porting through a radial passage formed in the piston valve member.

10. A pump system as claimed in claim **1**, said system including pump means having separate first and second intake ducts from which fluid is pumped into said first and second delivery passages respectively, said overspill ducting being connected to said first intake passage for delivering overspill fluid thereinto.

11. A pump system as claimed in claim **3**, wherein a flow restrictor is provided through which the fluid passing through the entry port flows.

12. A pump system as claimed in claim **11**, wherein the entry port and the main outlet port are axially spaced and the valve member and the valve bore together form said flow restrictor, said restrictor extending between the entry and main outlet ports, and the pressure at the upstream end of the flow restrictor being applied to an end of the valve member in opposition to the spring force.

13. A pump system as claimed in claim **1**, wherein the piston valve and transfer valve comprise respective piston valve members mounted in respective bores and are loaded into an initial stop position by respective springs disposed in spring chambers in the respective bores.

14. A pump system as claimed in claim **13**, wherein an entry port for the first fluid flow and a main outlet port open to the control valve bore, and said ports are in permanently open communication with each other, the pressure of the fluid flowing through the entry port being applied to the control valve member in opposition to the forst of its spring.

15. A pump system as claimed in claim **14**, wherein a pressure transmission passage is provided for transmitting to the spring chamber of the transfer valve a pressure of the fluid flowing through the entry port in the control valve bore.

16. A pump system as claimed in claim **14**, comprising a restrictor through which the fluid from the main outlet port is passed, and a duct extending between the downstream side of the restrictor and said spring chamber in the control valve bore.

17. A pump system as claimed in claim **16**, wherein the second delivery passage and an outlet port leading to the first delivery passage open at axially spaced locations to the transfer valve bore and, in the initial position of the transfer valve member, are in communication with each other only through an axially extending passage in the valve member, and wherein the overspill porting opens to the transfer valve bore at an axial location between the second delivery passage and said outlet port leading to the first delivery passage, the transfer valve member having a land which permanently obstructs communication between the second delivery passage and the overspill porting except by way of said axial passage in the transfer valve member,

said land obstructing communication between the outlet port and the overspill ducting in the initial position of the transfer valve member but opening communication as the transfer valve member is moved against the force of its spring.

18. A pump system as claimed in claim 17, wherein the control valve member has a portion which blanks off communication between the entry port for the first fluid flow and the overspill porting, and said pressure of the fluid which flows through the entry port urges the transfer valve member in a direction progressively to move said portion to bring the entry port into communication with the overspill porting.

19. A pump system as claimed in claim 18, wherein a spring-loaded relief valve is provided for limiting the pressure in the spring chamber of the control valve member.

20. A pump system as claimed in claim 19, wherein the relief valve is mounted in a passage extending axially along the control valve member, which passage is in permanently open communication with the overspill porting through a radial passage formed in the control valve member.

21. A pump system as claimed in claim 1, wherein an orifice is provided in the transfer passage for limiting flow of the fluid between the first and second delivery passages.

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